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EXPERIMENTAL INVESTIGATION OF AN ACCELEROMETER-CONTROLLED AUTOMATIC BRAKING SYSTEM

by Robert C. Dreher, Robert K. Sleeper, and John R. Nayadley, Sr.

Langley Research Center
Hampton, Va. 23365

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16. Abstract

An investigation was made to determine the feasibility of an automatic braking system for arresting the motion of an airplane by sensing and controlling braked wheel decelerations. The system was tested on a rotating drum dynamometer by using an automotive tire, wheel, and disk-brake assembly under conditions which included two tire loadings, wet and dry surfaces, and a range of ground speeds up to 70 knots. The controlling parameters were the rates at which brake pressure was applied and released and the Command Deceleration Level which governed the wheel deceleration by controlling the brake operation. Limited tests were also made with the automatic braking system installed on a ground vehicle in an effort to provide a more realistic proof of its feasibility.

The results of this investigation indicate that a braking system which utilizes wheel decelerations as the control variable to restrict tire slip is feasible and capable of adapting to rapidly changing surface conditions. For the test conditions which ultimately led to a locked-wheel skid, the system was observed to delay wheel lockup by controlling the braking action and thus has the potential of providing time for corrective action. In addition, because it permits little slip in the tire-ground interface, the accelerometer-controlled automatic braking system should afford the tire with good steering capability during normal braking operations.

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SUMMARY

An investigation was made to determine the feasibility of an automatic braking system for arresting the motion of an airplane by sensing and controlling braked wheel decelerations. The system was tested on a rotating drum dynamometer by using an automotive tire, wheel, and disk-brake assembly under conditions which included two tire loadings, wet and dry surfaces, and a range of ground speeds up to 70 knots. The controlling parameters were the rates at which brake pressure was applied and released and the Command Deceleration Level which governed the wheel deceleration by controlling the brake operation. Limited tests were also made with the automatic braking system installed on a ground vehicle in an effort to provide a more realistic proof of its feasibility.

The results of this investigation indicate that a braking system which utilizes wheel decelerations as the control variable to restrict tire slip is feasible and capable of adapting to rapidly changing surface conditions. For the test conditions which ultimately led to a locked-wheel skid, the system was observed to delay wheel lockup by controlling the braking action and thus has the potential of providing time for corrective action. In addition, because it permits little slip in the tire-ground interface, the accelerometer-controlled automatic braking system should afford the tire with good steering capability during normal braking operations.

INTRODUCTION

The operation of most airplanes requires large amounts of energy to be dissipated through wheel braking in landings and aborted take-offs. Most commercial and military airplanes are arrested by braking systems designed to absorb this energy and, at the same time, to avoid the problems associated with locked-wheel skidding. Locked-wheel skids cause excessively worn or ruptured tires, increased stopping distances, and complete loss in steering capability. Considerable effort has been directed in recent years toward improving these antiskid braking control systems, but no system has yet been devised which completely avoids locked-wheel skidding or assures good airplane stopping and directional control performance on low friction surfaces. The number of incidents where the aircraft has left either the end or the side of a wet runway attests to this fact.

A fundamental problem associated with designing an antiskid braking system stems from a lack of understanding of the behavior of a braked tire. Meaningful experiments to obtain this knowledge are difficult to devise; however, some insight into the tire characteristics has been gained from limited braking tests, most of which have been conducted under steady-state conditions (for instance, refs. 1 and 2). These tests have shown that a tire exhibits a maximum braking force at relatively low levels of wheel slip and an unstable braking behavior at higher levels since the wheel tends to lock up. These tests have also shown that an appreciable loss in cornering capability is experienced at only moderate levels of wheel slip.

Another antiskid design problem can arise from the method used to derive the controlling variable. Most currently operational antiskid braking systems regulate brake pressure to seek the maximum braking force. These systems generally use sensing techniques which can introduce control time lags that, under certain conditions, may contribute either to a locked-wheel skid or to operations for extended periods of time at levels of slip where the cornering force is substantially reduced.

In the interest of improving antiskid braking systems, an analog computer study was conducted in reference 3 to explore the phenomena which affect the basic operation of skid control systems. The braking control system used in the study derived the control variable directly from a wheel angular accelerometer and sought to maintain braking below some preselected low wheel deceleration level without trying to seek the maximum coefficient of friction. The accelerometer control introduced little time lag in the system and provided close control of the braking action. The results of the study indicated that a braking force could be developed which produced measurable wheel slip but, because of the tire elastic characteristics, no appreciable tire slip with respect to the ground.

The objective of this report is to present the results of an experimental investigation to determine the feasibility of the braking concept explored analytically in reference 3. To meet this objective a braking system using this concept was applied to an automotive wheel and tire assembly equipped with disk brakes, and the braking behavior of the system was investigated on a rotating drum dynamometer. Wet and dry surface conditions were investigated on the dynamometer at surface speeds up to 70 knots under two wheel loading conditions. In addition, the system was installed and operationally tested on a ground vehicle having drum-type brakes. The basic hydraulic and electronic components used in the system described in this report are similar to those of many systems now in use but differences arise in the concept of control, dynamic response, amplification levels, and circuit constant settings.

SYMBOLS

Values are given in both SI and U.S. Customary Units. The measurements and calculations were made in U.S. Customary Units.

F	normal load
g	acceleration due to gravity
p	brake pressure
R	radius of surface of drum dynamometer
W	weight of drum dynamometer
ϵ	error signal
$\ddot{\theta}_C$	Command Deceleration Level
$\ddot{\theta}_D$	deceleration of drum dynamometer
$\ddot{\theta}_W$	wheel deceleration
μ	coefficient of friction

AUTOMATIC BRAKING SYSTEMS

Braked Tire Considerations

In order for an antiskid braking system to safely arrest vehicle motion, adequate braking and cornering forces must be developed by the tire. In references 1 and 2, these characteristics have been found to vary with wheel slip, which is an apparent slip as measured about the wheel axis of rotation and includes tire slip and tire elasticity. Typical results from reference 2 are shown in figure 1 which presents the variation of dry braking and cornering coefficients of friction with wheel slip ratio. Wheel slip ratio is defined as the ratio of the difference in the free-rolling and instantaneous wheel angular velocities to the free-rolling wheel angular velocity. Figure 1 shows that with increasing slip ratio the braking friction coefficient rapidly increases to a maximum at a relatively low wheel slip ratio and then slowly decreases until the wheel locks up. The cornering friction coefficient is shown to be the greatest when the wheel slip ratio is zero and to sharply decrease as the slip ratio increases. It is important to note that the cornering capability approaches zero as a locked-wheel skid occurs.

Some Antiskid Braking Background

Because wheel lockup occurs when the applied brake pressure produces a braking torque which exceeds the torque developed by the force at the tire-surface interface, antiskid systems control braking by systematically reducing and reapplying brake pressure. The state of the braked wheel is determined by comparing instantaneous wheel angular speed with some reference angular speed.

One of the earliest antiskid braking systems, described in reference 4, was a mechanical design which utilized an inertia flywheel to provide a reference velocity for braking action. The author reported that this braking system had long been used successfully on railway cars to prevent wheel flat spots. A similar version was applied to airplane operations but lacked the fast response to changing surface conditions encountered by airplanes. Later antiskid braking systems determined wheel speeds and controlled the braking action electronically. Although a few antiskid braking systems seek to control wheel slip, most attempts to control the rate of velocity decrease of the wheel and both types of control require a reliable measure of a braked wheel speed to determine corrective action. Such measurements are difficult to obtain even with today's sophisticated systems. Common techniques employed in determining wheel speeds are dc voltage generators and rate counters used in conjunction with magnetically induced pulsing schemes. Wheel deceleration is computed from the velocity signal. Sources of system performance degradation may be attributed to control-system lags introduced by deriving deceleration from velocity and by the slower velocity determination techniques.

Accelerometer-Controlled Braking System

The basic elements of the accelerometer-controlled braking system of this paper are shown schematically in figure 2 and consist of a wheel angular accelerometer, a high-speed solenoid-operated brake control valve, adjustable inlet and outlet orifices, a set of disk brakes installed on a wheel and tire assembly, and the electronic circuitry to control the valve in an on-off manner. The function of the accelerometer is to monitor the wheel behavior during braking so that, when the wheel deceleration $\ddot{\theta}_W$ exceeds the preselected or threshold Command Deceleration Level $\ddot{\theta}_c$, a signal from the electronic control circuitry commands the control valve to release brakes until acceptable levels of deceleration (below $\ddot{\theta}_c$) are once more attained. Brake pressure is applied as long as the wheel deceleration remains below $\ddot{\theta}_c$. The Command Deceleration Level (CDL), which is expressed in terms of deceleration and varied by a potentiometer in the electronic circuitry, offers a convenient means for controlling the wheel deceleration. The adjustable orifices are used to optimize the action of the brakes by controlling the rate at which brake hydraulic pressure is allowed to rise and decay. The braking action with this circuitry consists of cyclic application and release of brakes until the wheel stops rotating. For this investigation, no low-speed brake release was incorporated in the circuitry.

APPARATUS AND TESTS

The accelerometer-controlled automatic braking system was installed and tested on a wheel and tire assembly driven by a rotating drum dynamometer and also on one wheel of a ground vehicle. Most of the testing of the system was conducted on the dynamometer where test conditions could readily be measured and controlled. However, testing of the system on a ground vehicle was considered essential to verify that it could operate satisfactorily under more natural conditions encountered by wheeled vehicles.

Drum Dynamometer

The test wheel and tire installed on the drum dynamometer are shown in figure 3. The test wheel was constrained from all motion except that normal to the axis of the drum. A pneumatically actuated piston was used to position the wheel along this axis and force the tire against the drum.

The drum was capable of being driven up to a design speed of 90 knots at the drum surface by a tire pressed against the drum opposite the test wheel and mounted on the shaft of a 15 kW (20 hp) shunt-wound electric motor. During testing, the driving tire was withdrawn so that the test wheel and drum could spin freely. The drum surface had a radius of 0.835 m (32.88 in.) and was rough machined to an rms value of 6.35 μm (250 μin.). Water was sprayed directly in front of the tire when investigating wet surface conditions. In order to simplify data reduction, the dynamometer was designed so that the weight or inertia could be varied by using ring-shaped plates without altering the radius of gyration which was located at the tire-drum surface.

The instrumentation on the dynamometer consisted of dc voltage generators to measure the angular velocities of the drum and test wheel and a load cell to measure the magnitude of the normal load on the tire. A pressure transducer installed in the brake pressure line measured the brake pressure. The output of an angular accelerometer mounted on the wheel axle was taken from the rotating wheel by means of slip rings and was used as the controlling variable for the braking system.

The nominal drum weight for the basic tests was 4980 N (1120 lb), and to maintain the kinetic energy of the drum consistent with that of a wheeled surface vehicle, an attempt was made to make the normal load equal to the drum weight. However, since the normal load was adjusted under static conditions, some variations in the load occurred during the test which may be attributed to an interaction between the axle motion in the normal direction and the load-producing mechanism. Windage and bearing friction of the rotating drum were very small and were neglected in these tests.

Test Procedure on Dynamometer

The automatic braking system was applied to the conventional automotive wheel and tire and disk-brake assembly shown mounted on the drum dynamometer in figure 4. The tire and wheel had a weight of 230 N (51.9 lb) and a mass moment of inertia of 1.445 kg-m^2 (12.78 lbf-in-sec^2). A 8.55 × 15 bias-ply automobile tire, having a load range B and 4-ply rating, was inflated to 21 N/cm^2 (30 psi) with an undeflected radius of 0.37 m (1.21 ft).

The automatic braking system utilized the electronic circuitry described in the appendix to actuate a normally open solenoid valve in an on-off manner. Maximum system pressure was 580 N/cm^2 (840 psi), and with the valve in the brake-on position, the variable orifice controlled the rate of pressure rise under static conditions over a range which extended from 210 N/cm^2/sec to 5400 N/cm^2/sec (300 psi/sec to 7800 psi/sec). With the valve in the brake-off position, the rate of pressure decay was varied from 5900 N/cm^2/sec to 16 500 N/cm^2/sec (8500 psi/sec to 24 000 psi/sec).

Prior to each test, the inlet and outlet orifices were adjusted as desired, the Command Deceleration Level was set, and the normal load was applied. The test was performed by driving the drum and test wheel up to the desired speed, removing the drive wheel, and actuating the braking system. Parameters of the braking system which were measured and recorded on an oscillograph consisted of the valve signal, brake pressure, wheel acceleration and velocity, normal load, and drum velocity.

Test Procedure on Ground Vehicle

The ground vehicle used in the test of the automatic braking system was the automobile shown in figure 5. Automatic braking was incorporated on the drum-type brakes of the left front wheel with the brakes on the other three wheels inoperative. Because of the braking arrangement, the automobile speed at brake application was limited to 12 knots.

The fully equipped vehicle weighed 18 950 N (4260 lb) and the weight supported by the braked wheel under static conditions was 4310 N (970 lb). Identical valves and electronic logic of the dynamometer-tested model were installed and a pressure cylinder added to the vehicle to provide a hydraulic supply pressure of 550 N/cm^2 (800 psi). Approximate rates of pressure rise and decay under static condition used for the vehicle were 5100 N/cm^2/sec (7400 psi/sec) and 8270 N/cm^2/sec (12 000 psi/sec), respectively.

The ground vehicle tests were conducted by accelerating the vehicle to the braking speed, placing the transmission in neutral, and braking the vehicle to a stop. The behavior of the system was evaluated from oscillograph records showing brake pressure, angular acceleration of the wheel, and braked and unbraked wheel velocities.

SYSTEM CHARACTERISTICS

The characteristics of the accelerometer-controlled braking system were evaluated from time histories of a nominally configured system during tests on a rotating drum dynamometer, where the effect of variation in the system control parameters on the braking response could be readily determined. The control parameters, which consisted of the Command Deceleration Level and the rates of brake pressure application and release, were adjusted for the nominal configuration to produce a deceleration rate of 1/2g at the drum surface under dry conditions.

Basic Features

A time history of the nominal system operating on the dynamometer, first under dry then under wet conditions, is reproduced in figure 6. Shown in the figure are record traces of wheel angular acceleration, normal load, brake pressure, drum and wheel angular velocities, and the valve signal. An upward displacement of the traces except the wheel angular acceleration denotes increasing values. The valve was either off or on where the upper and lower positions of the signal trace denoted the respective states. For the test illustrated in this figure, the drum weight was 4980 N (1120 lb), the normal load was 5470 N (1230 lb), the initial speed of the drum was 60 knots, and the Command Deceleration Level was set at 70 rad/sec^2.

The initiation of the test is recognized by the shift of the valve signal trace from the pressure-off position and the record shows that braking was applied for about 0.25 sec before the deceleration of the wheel was sufficient to command brake pressure release. On the dry surface the pressure increased to about 330 N/cm^2 (480 psi) before dropping to zero. The wheel acceleration trace shows brief periods of increasing deceleration while the pressure was increased and an almost instantaneous acceleration when pressure was released. The drum velocity, which corresponds to vehicle velocity, decreased smoothly at an angular rate of 5 rad/sec^2 and corresponded to a surface deceleration of 4.2 m/sec^2 (13.7 ft/sec^2). The trace for the wheel velocity exhibited uniform decrements which correlated in time with the pressure surges and visually displays the instantaneous slip between the wheel and drum. The effective deceleration of the wheel was only 11.4 rad/sec^2 although the wheel developed as much as 70 rad/sec^2 instantaneous deceleration.

After the system had operated about 2.4 sec on the dry surface, water was sprayed on the dynamometer drum. The behavior of the system on the wet surface is shown to be similar to that under dry conditions. However, figure 6 shows that under wet conditions the valve signal was in the pressure-off position for slightly longer periods of time during each cycle and the brake pressure did not reach the levels obtained under dry conditions. These lower brake pressure levels may be attributed to the lower brake torque required to produce the preselected Command Deceleration Level on the more slippery wet surface. The reduction in braking friction coefficient is reflected in the drum surface deceleration which was reduced to 2.2 m/sec^2 (7.1 ft/sec^2) and a corresponding decrease in the average wheel deceleration to 6 rad/sec^2.

The effective coefficient of friction developed by the system may be derived from the drum deceleration with the aid of the following relationship:

$$\mu = \frac{WR}{gF} \ddot{\theta}_D$$

From this equation, the effective coefficient of friction obtained for the test shown in figure 6 was 0.39 on the dry surface and 0.20 on the wet surface. Subsequent tests at other initial speeds up to 70 knots showed that the effective coefficient of friction did not change significantly for the wet surface and decreased only slightly with increasing speed on the dry surface of the dynamometer.

Figure 6 also shows that the normal load remained relatively steady at 5470 N (1230 lb), and on both the dry and wet surfaces the system cycled at approximately 10 cycles per second. The test as depicted in figure 6 indicates that the system is capable of adapting very readily from a dry steel friction surface to that of wet steel without allowing the wheel to lock up.

Further insight into the operation of the system may be gained from an examination of the wheel velocity trace, which exhibits quite uniform cyclic deviations from a velocity coincident with the drum rotation for both wet and dry surface conditions. Since the wheel slip ratio is derived from a ratio of the braked and unbraked wheel velocity deviations, figure 6 indicates that the slip ratio ranges from approximately 0.05 at the beginning of the test to 1.0 near the end. However, the tire must not be operating at these slip ratios or it would exhibit irregular deviations and would lock up. The tire behavior theory proposed in reference 3 attributed the wheel slip ratio below the maximum available coefficient of braking friction to tire torsional deformation. If this theory is true and can be applied, then there would be no appreciable tire slip occurring and, hence, most of the apparent wheel motion as shown in the figure must be due to tire flexibility.

Effect of Variations in Control Parameters

Basic tests of the system showed that small variations in the Command Deceleration Level and rates of change in brake pressure can significantly alter the behavior of the braking system. For example, a low Command Deceleration Level can reduce or completely eliminate braking action; whereas, a locked-wheel skid develops at the higher command levels or when the rate of brake pressure release is too slow.

Controlled wheel lockup.- To demonstrate the behavior of the system when the Command Deceleration Level is excessive, the Command Deceleration Level was increased from 70 rad/sec^2 to 125 rad/sec^2 and the resulting behavior is depicted in figure 7. The figure indicates that an insufficient brake release period now exists and the wheel ultimately locks up. As the wheel approached the locked condition, the system continued to control braking action for 3 sec before the wheel locked up. When compared with a 0.2-sec interval which occurred between brake application and wheel lockup without braking control, this system is shown to offer increased time for corrective action. It is interesting to note that after the wheel locked, the coefficient of friction developed was found to be 0.12.

Command Deceleration Level.- The Command Deceleration Level adjustment was effected by a potentiometer to provide a simple means for controlling the braking behavior. It has already been shown that an excessive Command Deceleration Level will cause the wheel to lock up in a controlled manner. To indicate the effect on the motion of a vehicle at the lower Command Deceleration Levels, the drum surface deceleration is displayed in figure 8 as a function of Command Deceleration Level where the maximum value is 70 rad/sec^2. Braking was initiated for this series of tests at a speed of 43 knots. On the dry surface, the drum deceleration is shown to increase linearly with Command Deceleration Level over the test range. However, on the wet surface, the deceleration does not increase appreciably after a Command Deceleration Level of approximately 30 rad/sec^2 is reached. At this value, the drum surface deceleration is approximately 2.1 m/sec^2 (7.0 ft/sec^2) and appears to be the maximum obtainable on this surface.

Brake pressure rates.- The adjustment of the rates of brake pressure rise and decay was found to be critical in providing a fast-acting braking system that did not lock up. Consequently, there was a need to identify these rates of change when the configuration was specified. However, because the rates of pressure change were difficult to assess during a test and varied with time, the rate of change of pressure obtained while the wheel was stationary has been used. To show how these rates were obtained, a typical pressure signature which describes the rates of brake pressure rise and decay when pressure was applied and released under static conditions is shown in figure 9. Straight lines were drawn along the pressure trace passing through a point midway between the pressure extremes, as the pressure increased and decayed. The slopes of these lines were used to define the hydraulic behavior of the braking system and could be varied by changing the settings on the inlet and outlet orifices. For the signature depicted in figure 9, the rate of pressure decay was 18 000 $N/cm^2/sec$ (26 100 psi/sec) and the rate of pressure rise was 4210 $N/cm^2/sec$ (6110 psi/sec). (A subsequent discussion of figures 10 and 11 will show the significance of these slopes of the pressure traces in the operation of this braking system.)

The valve signal trace shown in figure 9 denotes the voltage applied to the brake control solenoid valve. The lower position of the trace indicates that the valve has been deenergized and, since it is normally open, the pressure rises. Similarly, the upper position indicates that the valve has been energized and the pressure decays. The time duration between valve operation and pressure response illustrates the amount of lag, or operating time present in the system which may be attributed to lags, associated with the solenoid valve and the hydraulic system. In figure 9 a solenoid time lag of approximately 43 msec is indicated as determined from the time of signal removal to the first evidence of any pressure change (represented by the small spike on the pressure trace). The figure also shows a hydraulic time lag of approximately 140 msec before any appreciable pressure rise occurred. However, the time lag which must be minimized to prevent wheel lockup is associated with the pressure decay, which for the condition illustrated was approximately 14 msec as measured from the time of valve signal application to the beginning of pressure decay.

For the tests of figures 6 to 8, the rate of pressure rise was adjusted to 3720 $N/cm^2/sec$ (5400 psi/sec) and the rate of pressure decay was set for 14 480 $N/cm^2/sec$ (21 000 psi/sec). The results of tests to evaluate the effect on wheel deceleration attributed to variations in the rates of brake pressure rise and decay are presented in figure 10. Figure 10(a) summarizes the results obtained on a dry surface and figure 10(b) summarizes those obtained on a wet surface. On both surfaces the Command Deceleration Level was set at 70 rad/sec^2 and the initial braking speed was 30 knots. The rates of pressure rise and decay were arbitrarily restricted to five values each by using only five respective settings for the inlet and outlet orifices; however, figure 10 shows that these rates are not independent of each other. For example, the rate of pressure rise for one inlet orifice setting is shown to vary with changes in the outlet orifice setting. Similarly, the rate of brake pressure decay for one outlet orifice setting varies with changes in the inlet orifice. A partial explanation of these variations may be attributed to the difficulty in assigning slopes to the traces such as that of figure 9.

Effective wheel deceleration values were determined from the effective slope of the wheel velocity trace and are given adjacent to each symbol. The data of figure 10(a) indicate that greater wheel decelerations are developed on a dry surface when the pressure both rises and decays slowly. However, when the surface becomes more slippery (fig. 10(b)), pressure rates of brake decay which develop a large deceleration on a dry surface may cause wheel lockup. With pressure decay rates less than 12 065 $N/cm^2/sec$ (17 500 psi/sec), locked-wheel skidding developed on the wet surface as denoted by the flagged data points. At higher pressure decay rates, the wheel did not lock up and the effect of the rate of pressure rise on wheel deceleration appeared negligible for the decay rates considered. On this wet surface a maximum wheel deceleration of 8 rad/sec^2 was developed for the range of brake pressure rates studied.

The effect of the rates of rise and decay of brake pressure on drum decelerations (which simulate vehicle decelerations) is shown in figure 11. Each curve for the dry surface was plotted for a fixed outlet orifice setting and depicts the drum deceleration at various rates of pressure rise. The maximum drum surface deceleration on the wet surface was 2.7 m/sec^2 (9 ft/sec^2) and is shown independent of the pressure rise rate. As previously indicated, for brake pressure rates of decay less than approximately 12 065 $N/cm^2/sec$ (17 500 psi/sec), the system developed a locked-wheel skid on a wet surface. The data of figure 11 indicate that on a dry surface the brake pressure rates of rise and decay have a large effect on the magnitude of the drum or vehicle deceleration. For example, with a constant decay rate of 15 170 $N/cm^2/sec$ (22 000 psi/sec), the deceleration increased from 3.66 to 7.0 m/sec^2 (12 to 23 ft/sec^2) as the rise rate is decreased from 4650 to 344 $N/cm^2/sec$ (6750 to 500 psi/sec). The drum deceleration also increased if the rate of rise was held constant and the rate of decay decreased.

Effect of Increased Wheel Loading

Since aircraft must operate under various loading conditions, it was necessary to verify that the system could tolerate load changes. The operation of the system with increased vehicle weight is illustrated in figure 12 which shows a time history of a test with the same system settings shown in figure 6, except the static normal load and the drum weight have been increased to 7380 N (1660 lb). Larger pressure excursions and longer cycling periods were experienced for this system and, as found previously, a lower maximum pressure results when water was introduced on the drum surface.

A comparison of figures 12 and 6 shows that the excursions noted for normal load during the test are shown to be larger with the increased loading and may be due to an increased interaction between the centrifugal tire growth and the pneumatically operated piston producing the normal load.

Figure 12 shows that a drum surface deceleration of 4.26 m/sec^2 (14.0 ft/sec^2) was developed on the dry surface and 1.83 m/sec^2 (6.0 ft/sec^2) on the wet surface. Since these values compare favorably with 4.2 m/sec^2 (13.7 ft/sec^2) and 2.2 m/sec^2 (7.1 ft/sec^2) found for the dry and wet conditions, respectively, with the lighter weight, the system behavior does not appear to be seriously affected by weight changes.

SYSTEM APPLICATIONS

Incorporation Into Ground Vehicle

To verify that the accelerometer-controlled automatic braking system could operate satisfactorily in a ground vehicle, the system was installed on one wheel of an automobile with all other brakes disconnected, and limited qualitative testing was conducted. Time histories of the angular acceleration, braked and unbraked wheel angular velocities, and the brake pressure are shown in figure 13 as the ground vehicle traveling at a speed of 12 knots was braked to a stop on a dry surface. Vehicle speed was established by monitoring the angular velocity of an unbraked wheel.

When the time histories of this figure are compared with the time histories for the dry surface in figure 6, the behavior of the system installed in the automobile is found to be quite similar to that observed for the system on the dynamometer. The pressure trace for the ground vehicle shows oscillations about an operating pressure of 260 N/cm^2 (380 psi) which are probably characteristic of the hydraulic installation on the automobile; however, these higher frequency oscillations are not reflected in the braked wheel behavior. The unbraked wheel velocity is shown to have small oscillations which were not distinguishable in the dynamometer drum response. The cause of the oscillation is not known but part of the discrepancy may be attributed to the higher recorder gain used in the ground vehicle tests.

Since these time histories resemble those from the drum dynamometer and since the vehicle motion was arrested in a uniform manner, the test suggests that the system can satisfactorily be applied to a ground vehicle.

Potential Applications

As implied in the discussion of figure 1, the operating slip ratios of this system are less than those experienced by control systems which maximize the braking coefficient of friction. Since research (ref. 2) has demonstrated that the cornering forces developed by a tire vary inversely with wheel slip ratio, the braking control provided by this system should maintain a larger degree of steering or cornering capability during braking. In addition, a reduction in tire wear should result from operating at the lower slip ratios.

Differential braking, whereby the wheels on one side of the airplane are braked at a different level than on the opposite side, is an important source of steering for the airplane. Such braking can be readily implemented on this system by coupling the Command Deceleration Level potentiometer to foot pedals. For such an application, figure 8 may be interpreted as the effect of foot pedal displacement on drum surface deceleration. The data of this figure would indicate that on a dry surface the drum deceleration would increase linearly with simulated foot pedal displacement (Command Deceleration Level) and provide good differential braking. Although drum deceleration does not increase linearly over the entire range of simulated pedal displacement on the wet surface, some steering through differential braking would be available.

A form of system adaptability was illustrated for the system when its operation automatically adjusted from a dry to a wet surface condition. Since the Command Deceleration Level was a primary control parameter that was effected by changing a potentiometer, the system lends itself to further adaptability by inclusion of the system within a completely adaptive automatic braking system.

CONCLUDING REMARKS

An automatic braking concept which developed a low level of tire slip and employed an angular accelerometer to control wheel braking was evaluated on a rotating drum dynamometer under dry and wet surface conditions. Limited tests were also conducted with the system installed on a ground vehicle. The results from these tests indicate that such a braking concept is feasible for operations on surfaces of different slipperiness.

Wheel braking was primarily controlled by specifying a Command Deceleration Level which in turn controlled the application and release of brake pressure; however, the rates of brake pressure rise and decay were found to be critical in order to make the system respond quickly without locking up. The automatic braking system, when adjusted to develop adequate braking force, proved capable of rapidly adjusting to abruptly changing surface friction conditions under two tire loadings. For those conditions which ultimately led to a locked-wheel skid, the system was observed to control the braking action; this control delayed an immediate wheel lockup and thereby provided increased time to initiate corrective action.

Because the automatic braking system permits little tire slip during braking, it would appear that the system provides the potential for increased tire cornering capability and reduces tread wear during braking. Furthermore, the system appears capable of providing differential braking and lends itself for incorporation into completely adaptive braking systems.

Although this study indicates that an accelerometer-controlled automatic braking system is feasible, additional study would be valuable for its development. Further research is needed to verify that the cornering capability during braking is improved and to test the system on a ground vehicle at speeds more representative of those experienced by airplanes during braking.

Langley Research Center,
National Aeronautics and Space Administration,
Hampton, Va., September 19, 1972.

APPENDIX

ELECTRONIC CONTROL UNIT

A solid-state electronic control device was designed for the automatic braking system to utilize the signal from a strain-gage-type angular accelerometer to control the release and application of brake pressure by the cyclic actuation of a solenoid valve. As depicted in figure 14, the device basically compares the measured deceleration with a preselected Command Deceleration Level and when the measured deceleration exceeds the preset value, the solenoid valve is commanded to release the hydraulic brake pressure until the measured deceleration is again less than the command value.

A schematic diagram of the controlling device is shown in figure 14 and consists of a dc amplifier, Schmidt trigger circuit, two-stage beta multiplier, and a power transistor switch. Details of its features and a discussion of its operation follows. The dc amplifier provides a gain of approximately 90. The amplified accelerometer signal operates the Schmidt trigger circuitry which is composed of transistors Q_1 and Q_2 and associated resistors. The Schmidt trigger provides an almost instantaneous switching response to the error signal which is derived from the difference between the measured and preset deceleration levels. Transistors Q_3 and Q_4 provide current and gain and control the state of the power transistor Q_5. When Q_5 is in saturation, the solenoid is actuated by the current flowing through Q_5 and the solenoid coil.

The Command Deceleration Level which controls the electronic control unit and hence the complete braking system is determined by the settings of variable resistors R25 and R42.

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2. Byrdsong, Thomas A.: Investigation of the Effect of Wheel Braking on Side-Force Capability of a Pneumatic Tire. NASA TN D-4602, 1968.
3. Batterson, Sidney A.: A Study of the Dynamics of Airplane Braking Systems as Affected by Tire Elasticity and Brake Response. NASA TN D-3081, 1965.
4. Bent, Arthur J.: No More Skids For Airplanes! SAE J., vol. 58, Jan. 1950, pp. 63-66.

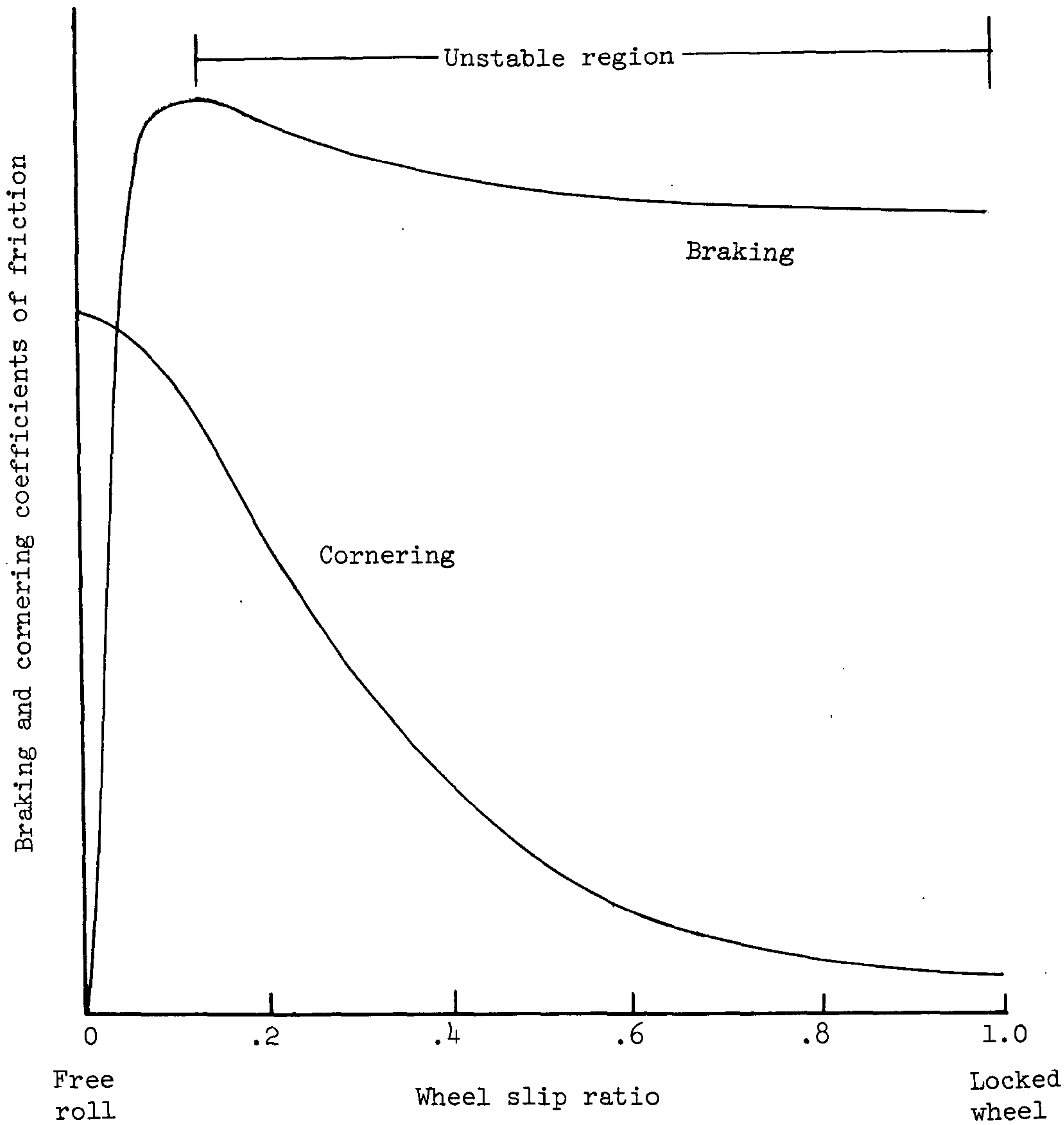

Figure 1.- Typical variation of coefficient of friction with wheel slip ratio.

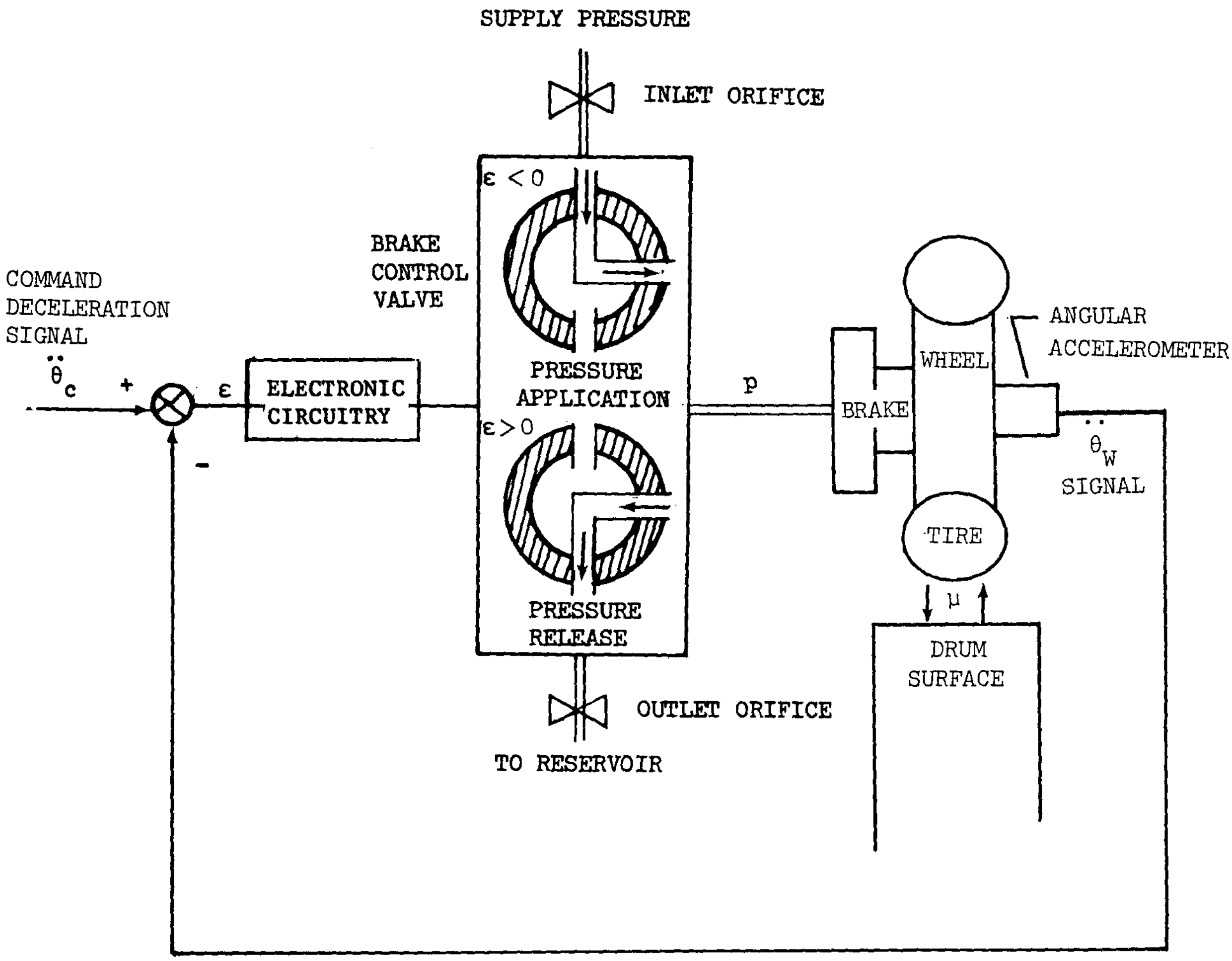

Figure 2.- Schematic diagram of accelerometer-controlled automatic braking system.

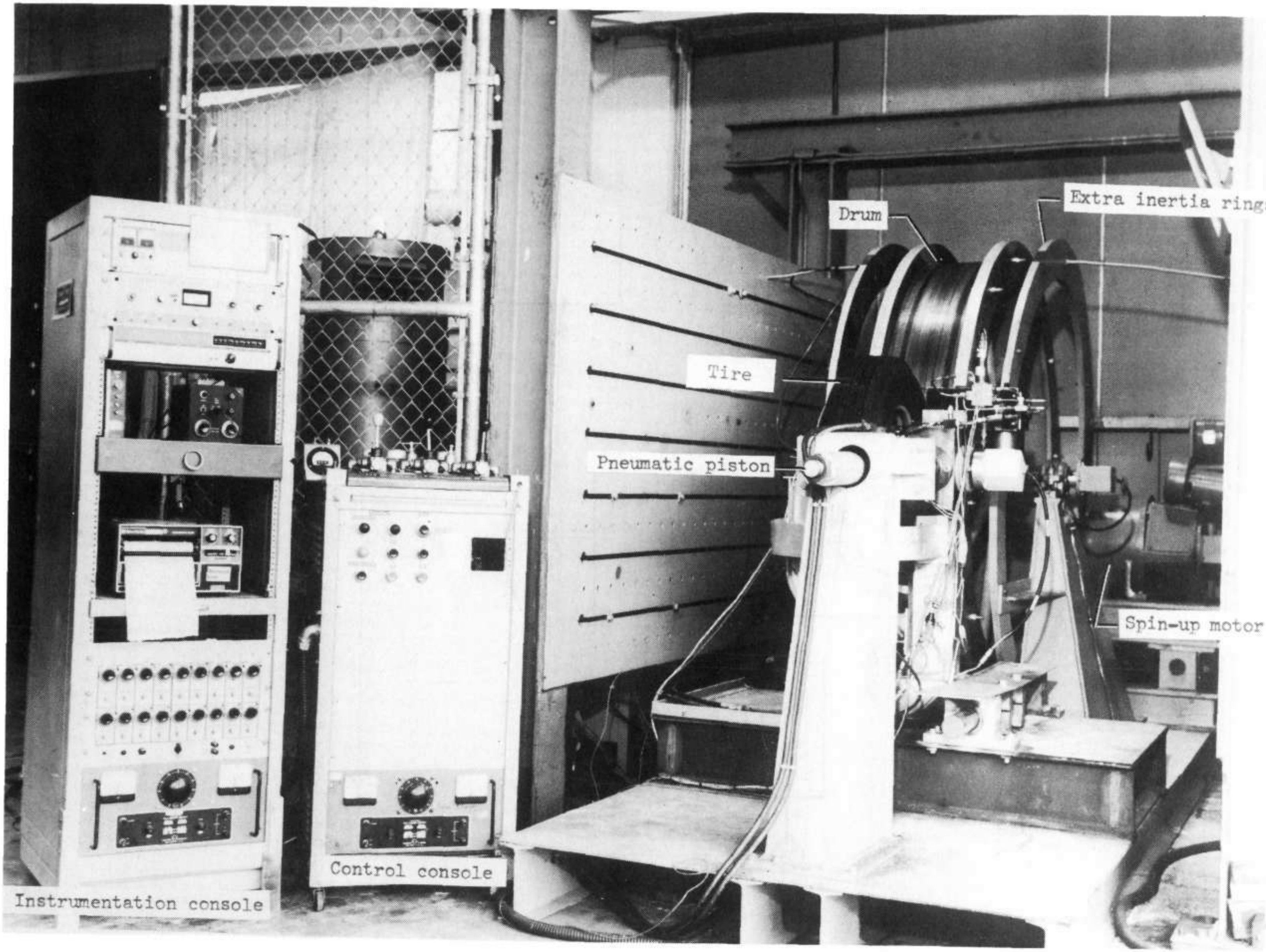

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Figure 3.- Drum dynamometer and associated equipment used in tests.

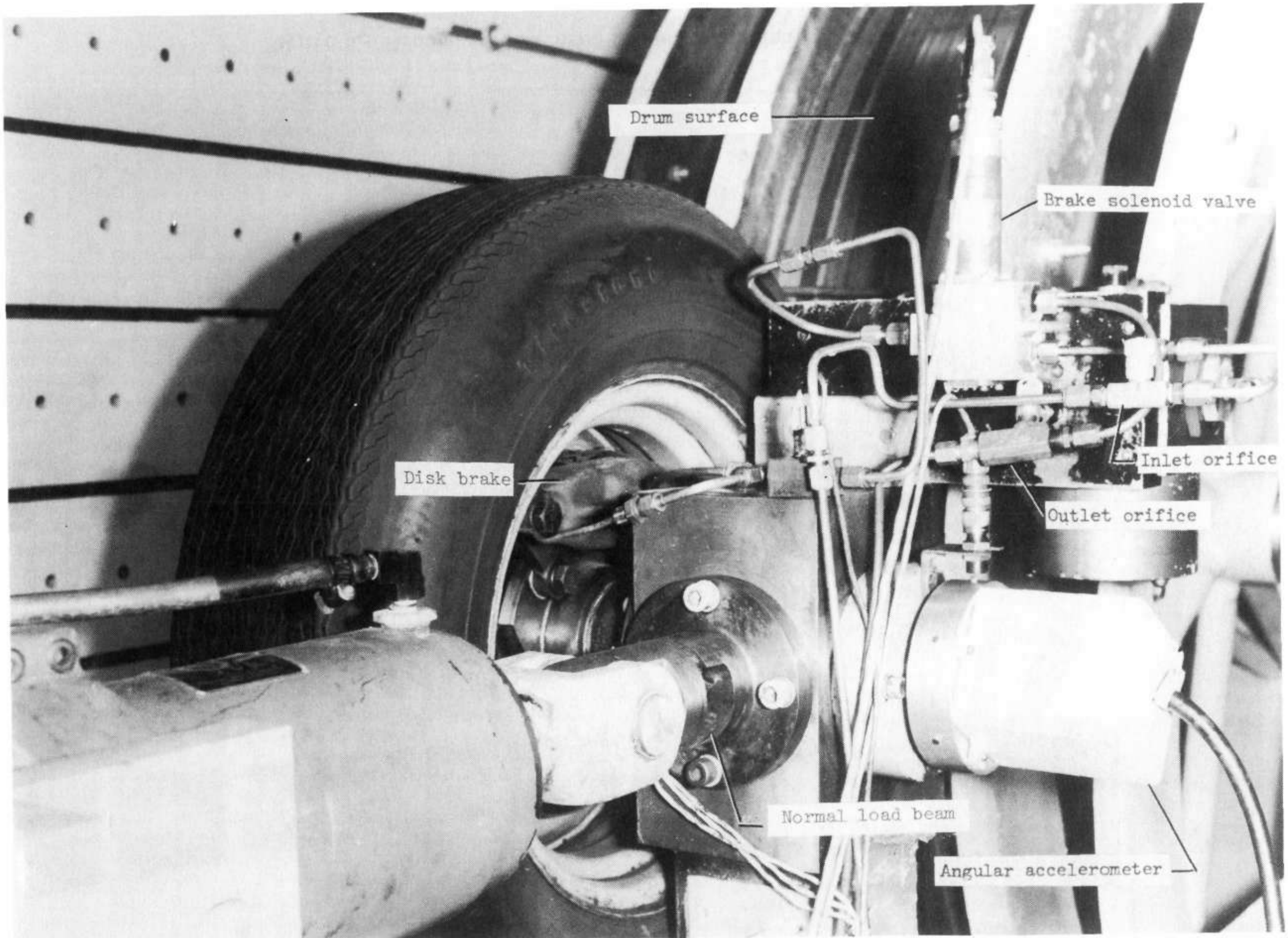

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Figure 4.- Components of automatic braking system.

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Figure 5.- Ground vehicle used in test of automatic braking system.

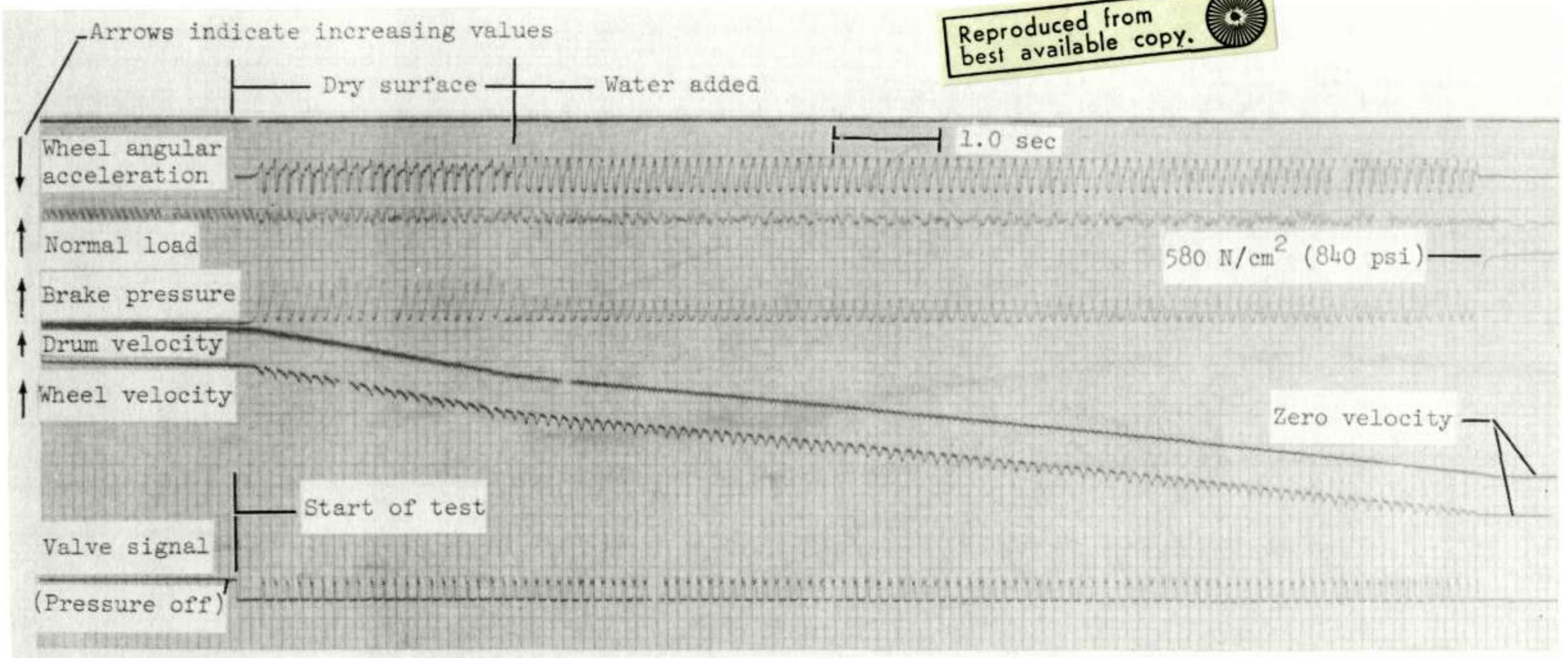

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Figure 6.- Oscillograph record of braking system operation on dry and wet surfaces. Initial speed, 60 knots; Command Deceleration Level, 70 rad/sec^2; normal load, 5470 N (1230 lb); drum weight, 4980 N (1120 lb).

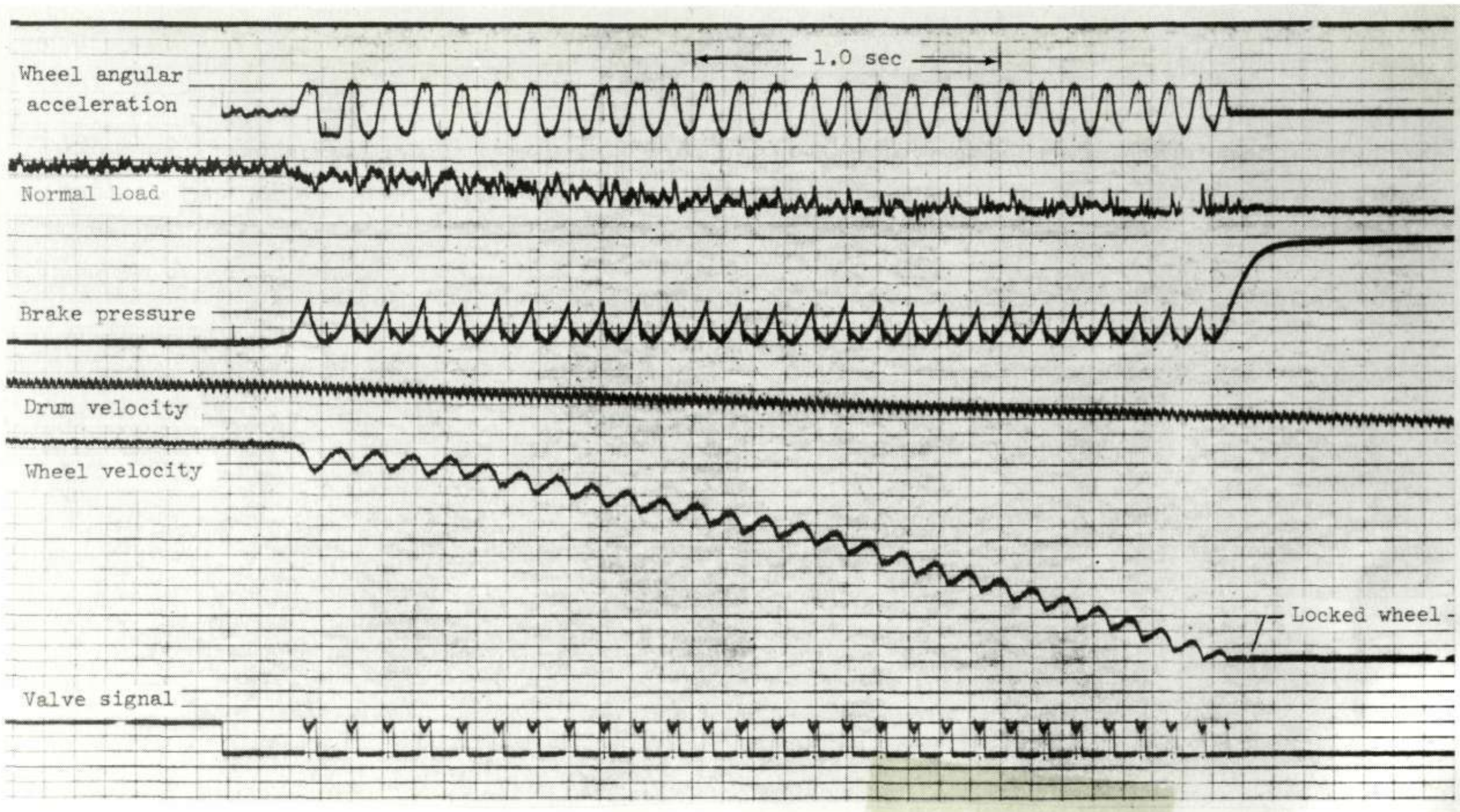


Figure 7.- Typical controlled wheel lockup on a wet surface. Initial speed, 60 knots; Command Deceleration Level, 125 rad/sec^2.

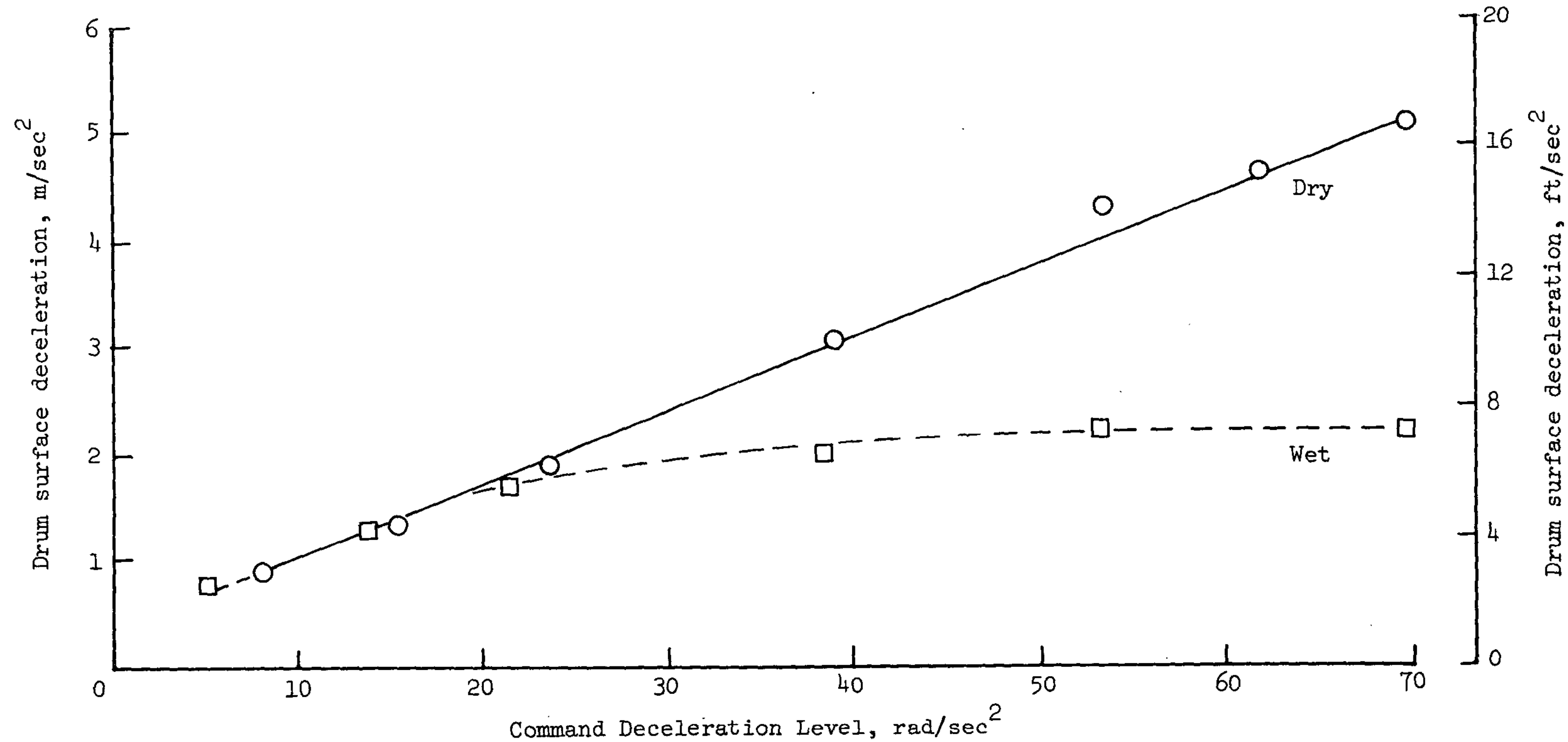


Figure 8.- Effect of Command Deceleration Level on drum deceleration on dry and wet surfaces.

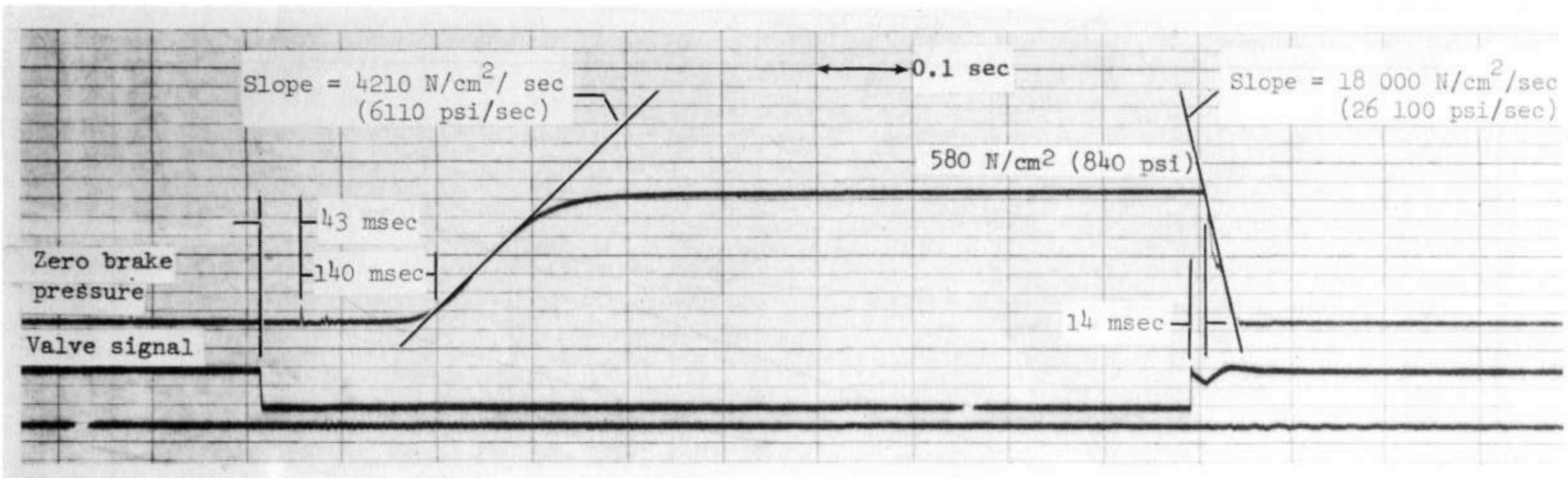
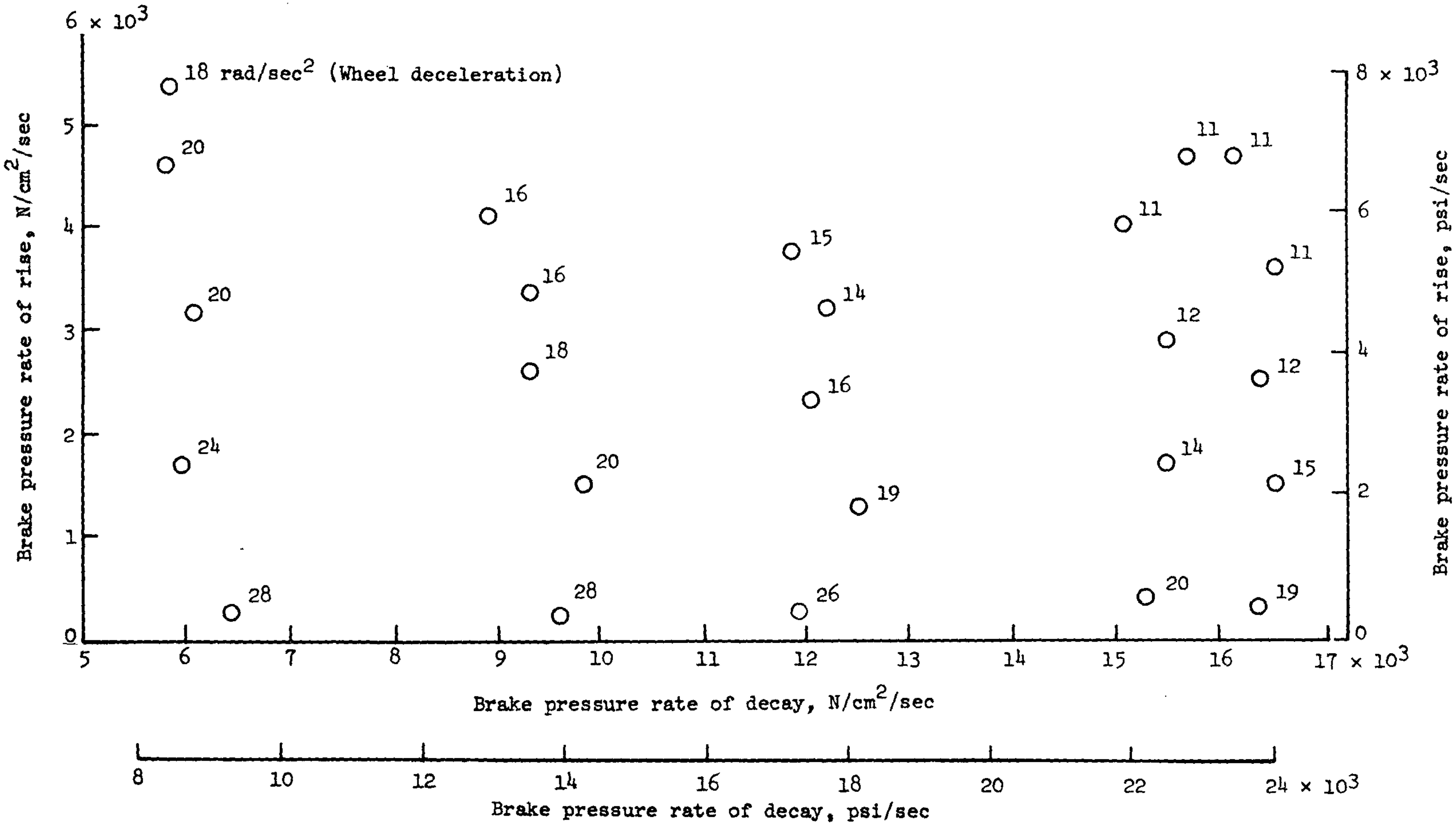
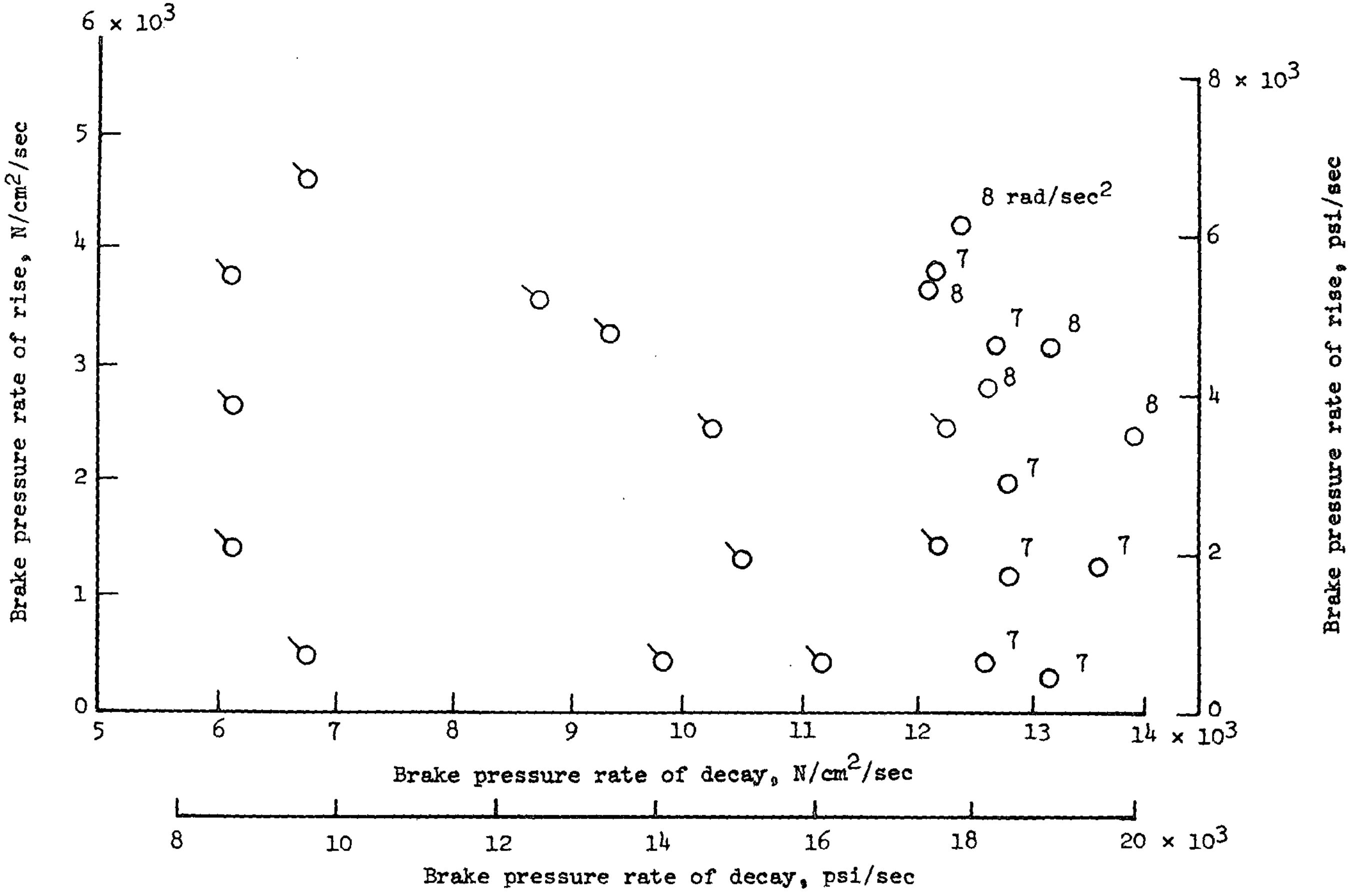


Figure 9.- Typical static-pressure signature of automatic braking system.



(a) Dry surface.

Figure 10.- Wheel decelerations obtained with various combinations of brake pressure rates of rise and decay with a Command Deceleration Level of 70 rad/sec^2.



(b) Wet surface. Flagged symbols denote wheel lockup.

Figure 10.- Concluded.

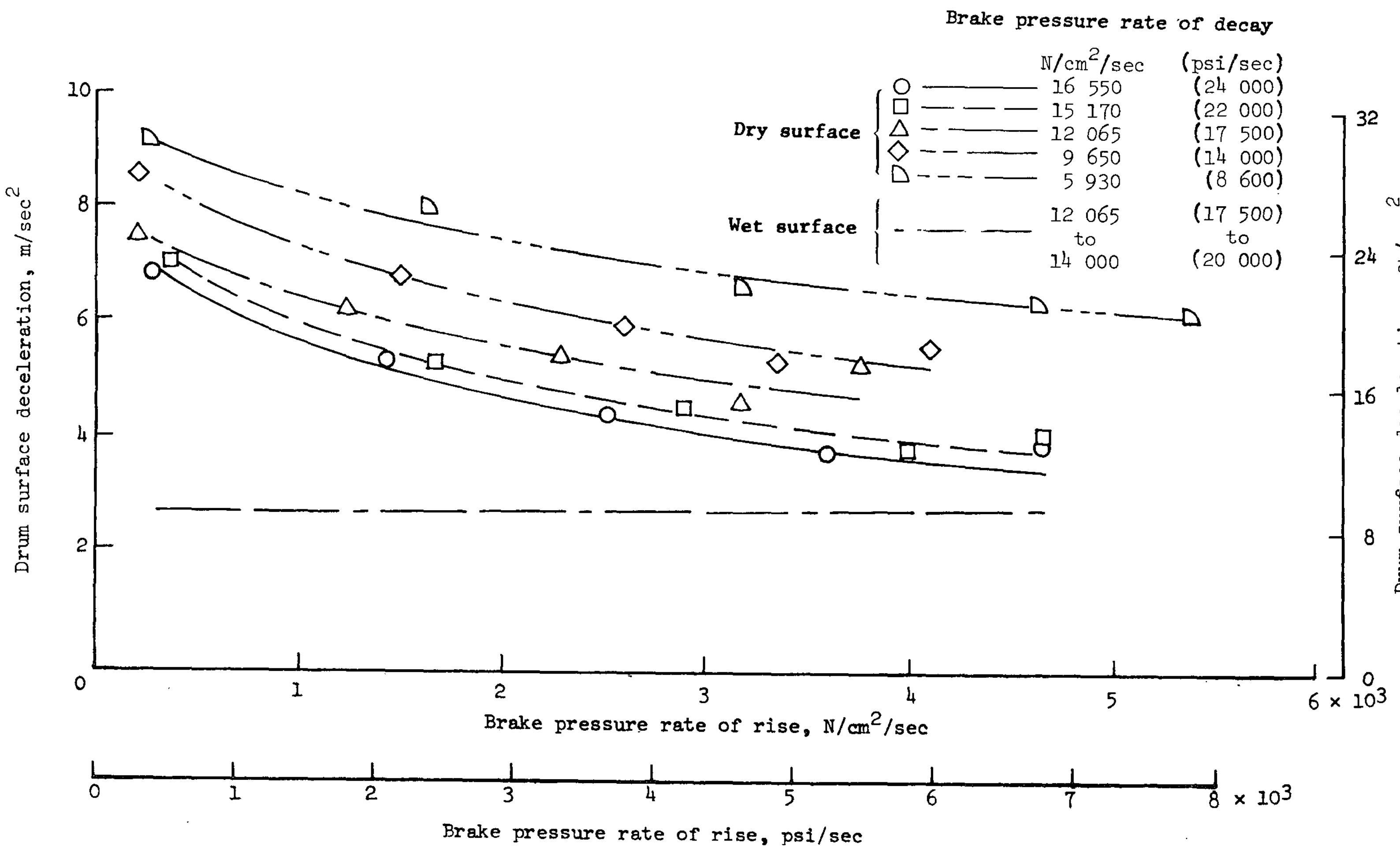


Figure 11.- Effect of brake pressure rate of rise on drum surface deceleration at various brake pressure rates of decay with a Command Deceleration Level of 70 rad/sec^2.

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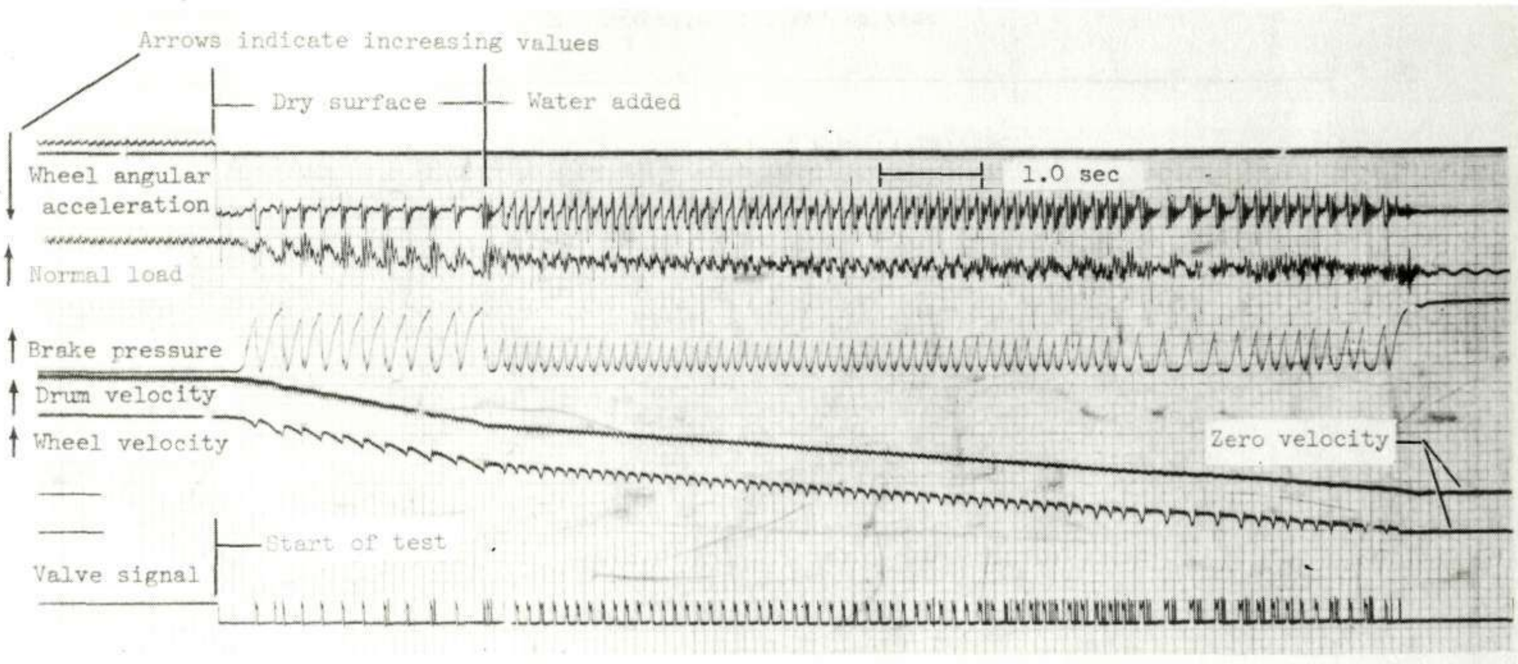

Figure 12.- Operation of automatic braking system with increased vehicle weight on dry and wet surfaces. Initial speed, 60 knots; Command Deceleration Level, 70 rad/sec^2; normal load and drum weight, 7380 N (1660 lb).

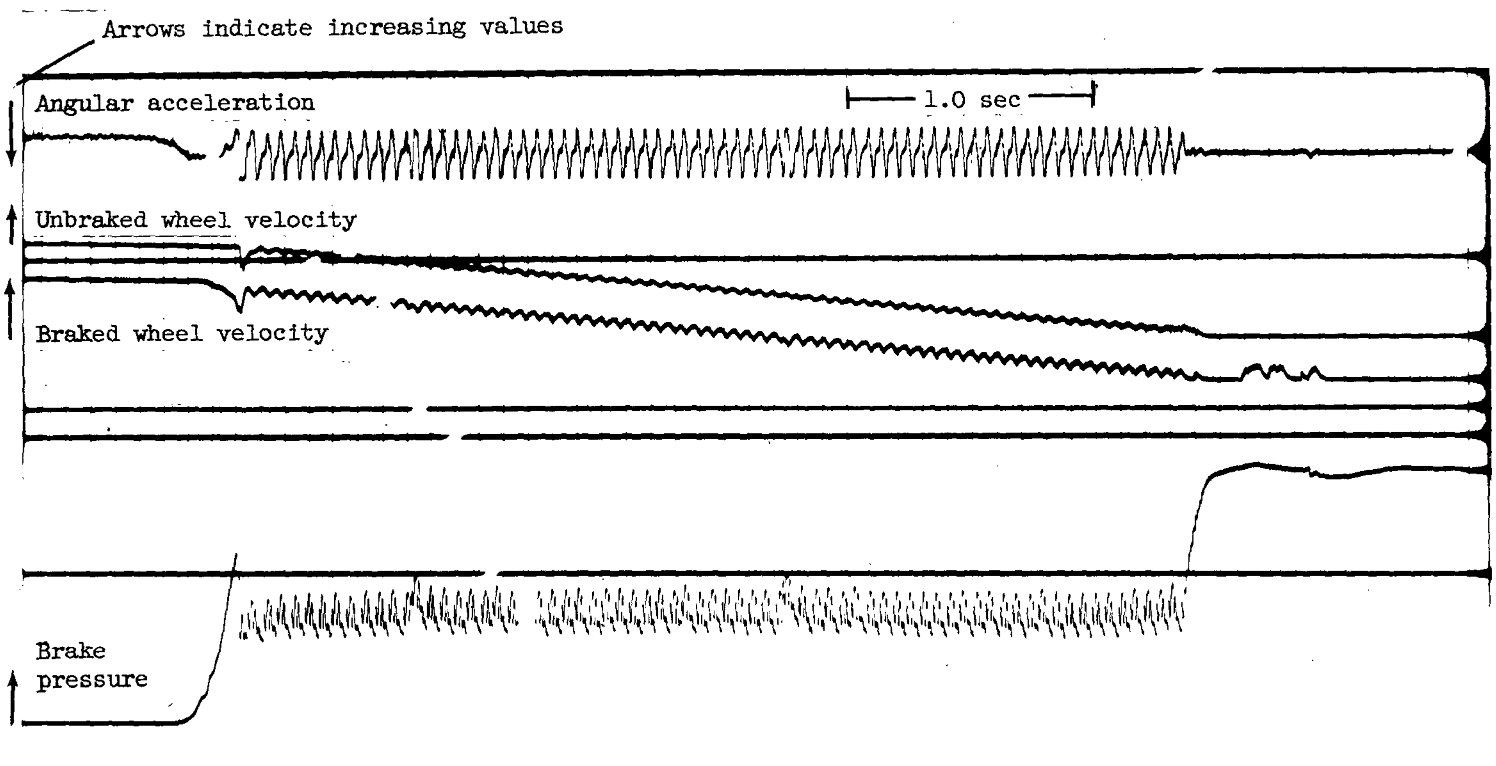

Figure 13.- Typical oscillograph record obtained with automatic braking system installed on ground vehicle.

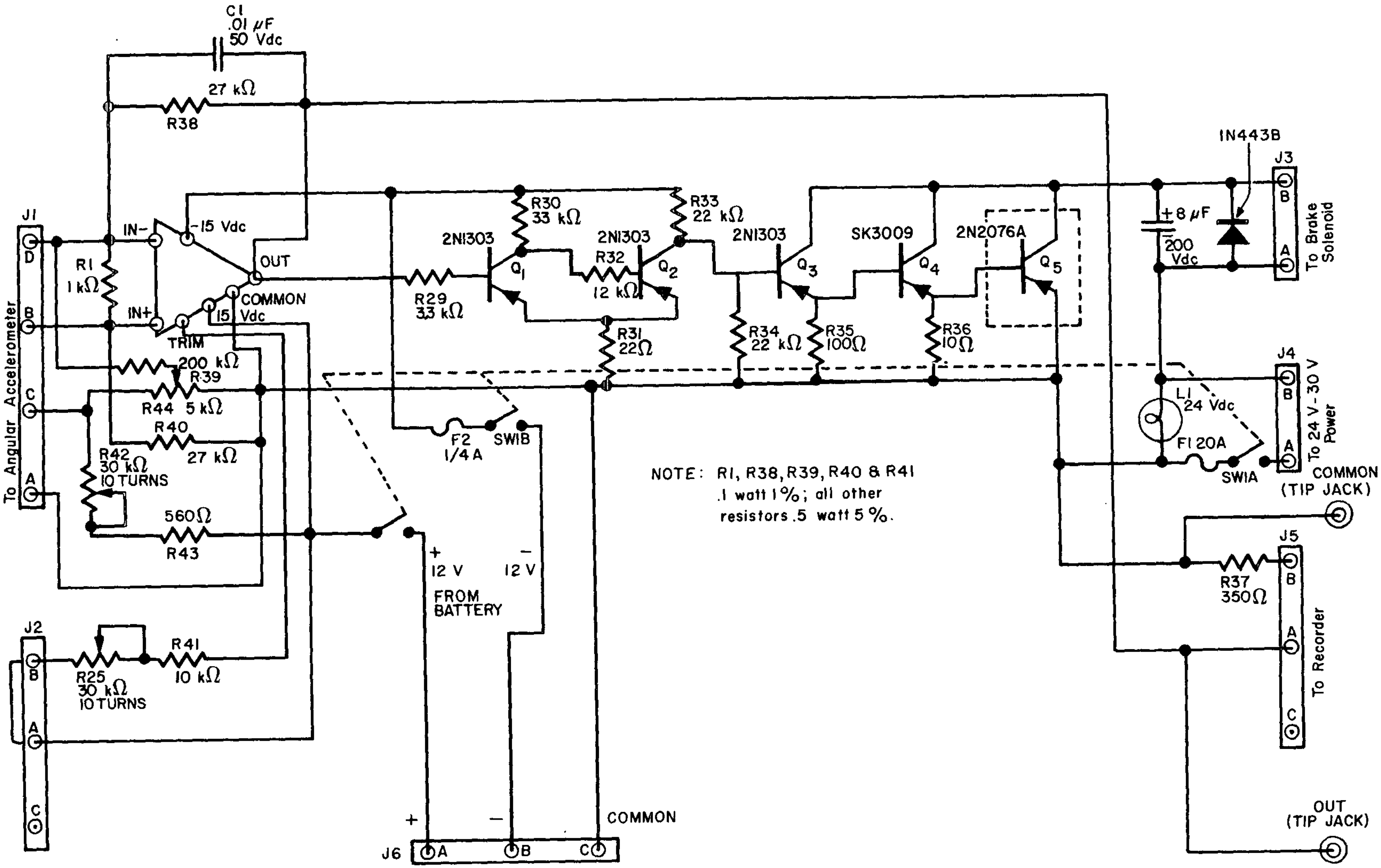

Figure 14.- Schematic of electronic control unit.

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